Damping structure and applications

The present invention relates to a damping structure and applications of such a damping structure.

A preferred application relates to the use of the damping structure to damp out vibrations of vibrating components such as the main gearbox of a rotary wing aircraft, particularly a helicopter, especially with a view to reducing the noise in the cockpit and/or in the passenger cabin of said aircraft.

It is known that, in a rotary wing aircraft, the acoustic spectra defined in the range between 20 Hz and 20 kHz are the result of the superposition of noises of various origins which can be grouped into two different groups depending on their spectral characteristics, namely pure sounds or narrowband noise and broadband noise.

In the known way, pure sound or narrowband noise occurs in particular, as appropriate:

- at the characteristic frequencies of the aircraft driveline;
- at the rotational frequencies of the rotor blades (main and tail) and at the harmonics of these frequencies;
- at the rotational frequencies of the blades of the compressors of the turbine engine units; and/or
- at the rotational frequencies of the blades of the blowers that cool the main gearbox or distribute cabin air and/or of electrical equipment, and at the harmonics of these frequencies,

while broadband noise comprises, in particular, as appropriate:

- the noise of the boundary layer that develops along the fuselage;
- the noise generated by the rotors;
- the noise of the air intake and nozzle flows;
- the engine noise; and/or
- the noise of the circuits that provide the cockpit

or the passenger cabin with climate control or heating.

All this noise is of course troublesome for the pilots and the passengers.

There are various known solutions for reducing such noise inside a rotary wing aircraft, particularly a helicopter.

The object of a first known solution is to reduce the level of vibration or the radiation of sources of noise and/or of the fuselage. To this end, various physical actions can be taken, particularly:

- reducing the vibration of the structure and/or of mechanical parts, by damping or modifying the stiffness or the mass;
- attenuating the acoustic transmission, by damping or modifying the stiffness or the mass;
- a double baffle system, using a space which may or may not be filled with absorbent material, between the radiating structure and soundproofing panels;
- acoustic absorption using fibrous or cellular materials; and
- acoustic absorption using Helmholtz resonators.

The first four physical actions listed above make it possible to reduce the overall noise level in a broad range of frequencies, but would lead to a significant and highly disadvantageous increase in mass. In addition, the obtained reduction in noise is not selective enough to eliminate the acoustic annoyance specific to the emergence of pure sounds.

By contrast, the fifth and final physical action listed above makes it possible effectively to reduce narrowband noise, but still only in a narrow band of frequencies, defined during design.

This first solution listed above and based on a passive treatment of the noise is therefore not completely effective, particularly in the case of narrowband noise generated by vibrational excitation.

A second known solution recommends creating passive soundproofing in the form of cladding panels

mounted in the cockpit or in the passenger cabin. These panels are designed according to the structural area that is to be treated and according to the spectrum of frequencies to be attenuated.

However, this second solution also has numerous disadvantages and, in particular:

- the noise reduction is limited, especially at low frequencies;
- the increase in mass is high, and may be several hundreds of kilograms in the case of a large-sized helicopter;
- there is a not insignificant loss of volume, especially when using thick panels with a view to improving the acoustic absorption effect; and
- there are acoustic leakages, particularly at the wiring holes and the joints between the panels.

In consequence, neither of these two known solutions listed above is satisfactory in reducing the annoyance caused by noise, particularly narrowband noise.

One of the objects of the present invention is to propose a solution that makes it possible to reduce such noise.

To this end, the present invention relates to a damping structure which is simple and inexpensive to produce, having numerous advantages and which can be used in various applications to deaden the vibrations generated by vibrating sources, particularly with a view to reducing noise, and to do so particularly in a rotary wing aircraft such as a helicopter.

To this end, said damping structure is notable, according to the invention, in that it has an internal cavity and comprises:

- an aggregate which comprises at least solid bodies in contact and which completely fills said internal cavity; and
- means for closing off the internal cavity and pressing said aggregate into said internal cavity.

Thus, when said structure is subjected to

vibration, these vibrations are transmitted to the (contacting) solid bodies of the aggregate, via the various points of contact. On crossing each of these points of contact, some of the vibrational energy is dissipated through friction so that said vibration is then damped quickly and effectively within said structure.

As a preference, said structure is elongate, for example in the form of a bar, and said internal cavity is formed longitudinally inside said elongate structure.

In the context of the present invention, said solid bodies, which are, for example, made of synthetic material, preferably beads, may be:

- either compact, their entire mass then being occupied by material;
- or hollow, which makes it possible to reduce the weight of said solid bodies and therefore also to reduce the weight of the structure.

In addition, according to the invention, said solid bodies may be made of different materials (synthetic material, metal, etc.) and/or have different shapes and/or different sizes (diameters).

It will be noted that:

- a difference in inertia of said solid bodies, due in particular to different sizes or densities; and/or
- a difference in stiffness of said solid bodies, due especially to different materials (for example a material which is not very rigid and intrinsically has very good damping properties or a material which is more rigid and intrinsically has less good damping properties),

give rise to a different movement under the effect of a vibrational excitation and therefore also to a different degree of damping. In consequence, through a suitable selection of these characteristics, it is possible to adjust and to optimize the damping performed by the damping structure according to the invention.

Furthermore, advantageously, said structure additionally comprises at least one internal partition, solid or pierced, of any shape, particularly tubular, which may or may not be secured to the wall of said structure and which is arranged inside said internal cavity.

This makes it possible to increase the area for exchange (friction) between the structure and the aggregate and therefore also the damping of the vibrations.

Furthermore, advantageously:

- in that said aggregate additionally comprises a viscous liquid filling the spaces between said solid bodies; and/or
- said means for closing off said internal cavity comprise a rigid plate which is constrained by an elastic element.

In addition to the aforementioned advantages, the damping structure according to the invention also has the following advantages:

- it can easily be produced and has a low manufacturing cost, particularly when the internal cavity already exists in the structure;
- it is of low mass (particularly when use is made of hollow solid bodies) by comparison with certain known damping means, such as the bonding of viscoelastic materials, which may or may not be constrained, onto the surface of the structure that is to be damped;
- the aggregate it contains is protected against external attack (fire, moisture, corrosive agents, etc.) by the structure itself;
- it is effective over a broad band of frequencies and this is true for various types of deformation (bending, tension-compression, torsion, etc.) of the structure;
- it is not subjected to phenomena of abrasion, corrosion or erosion, if an appropriate pair of materials is chosen for the wall of the structure

and for the aggregate, respectively; and

- it leads to no change in the lifespan of the components with which it is associated.

In one particular application, said structure may be produced in the form of a hollow pinion (of a gearbox or of any other mechanical device) which is filled with said aggregate according to the invention.

The present invention also relates to a suspension system for a gearbox of a rotary wing aircraft, particularly a helicopter.

According to the invention, said suspension system which comprises a number of suspension bars is noteworthy in that at least one of said suspension bars comprises a structure as mentioned hereinabove.

Thus, the equivalent damping of at least one of said bars is improved, which makes it possible effectively, in the cockpit and/or the passenger cabin of the aircraft, to reduce the noise propagated through solid which is transmitted by said treated bars.

The present invention also relates to two types of damping device using the aforementioned structure, for damping the vibration of any vibrating component, for example:

- a connecting rod;
- an engine;
- a gearbox; or
- a rotary part such as a compressor or a fan for example.

A first of these damping devices comprises a damping structure according to the invention which is arranged between the vibrating component and a support.

It will be noted that the structure used is rigid and can either be attached in an empty space or directly replace an element that already exists for performing other functions, particularly mechanical or structural functions, such as a connecting rod for example.

A second damping device for damping vibration of a vibrating component comprising at least one hollow

element, for example a suspension bar for suspending said vibrating component, is obtained by producing said element in the form of the aforementioned damping structure. In the context of the present invention, the cavity of this element may either be a cavity that already exists or a cavity made specifically for implementing the present invention.

This second damping device has the additional advantage of not increasing the space occupancy.

The figures of the appended drawing will make it easy to understand how the invention may be embodied. In these figures, identical references denote elements which are similar.

Figure 1 schematically shows a damping structure according to the invention.

Figures 2 and 3 show structures according to the invention comprising various types of solid bodies.

Figures 4 to 7 and 8 to 11 show, schematically, various embodiments of internal partitions of the structure according to the invention, in a longitudinal view and in a plan view, respectively.

Figure 12 schematically illustrates a mechanical breakdown of the structure according to the invention.

Figure 13 shows a preferred application of the structure according to the invention, relating to the suspension of the main gearbox of a helicopter.

The damping structure 1 according to the invention and depicted schematically in figure 1 is a mechanical element specified hereinbelow which, according to the invention, has an internal cavity 2 surrounded by walls 3, 4 forming a chamber 6 and opening via an opening 7.

According to the invention, said structure 1 comprises:

- an aggregate 8 which comprises solid bodies 9 in contact and which completely fills said internal cavity 2 although in order to simplify the drawing, solid bodies 9 have not been depicted in the whole

of the cavity 6 in figure 1; and

- means 10 for closing off the internal cavity 2 and pressing said aggregate 8 into said internal cavity 2 against said walls 3 and 4.

Thus, when the structure 1 is subjected to vibration, for example longitudinal vibration E or lateral vibration F, this vibration is transmitted via the walls 3 and 4 to the solid bodies 9 (which are in contact) of the aggregate 8 which is pressed, via the various points of contact. On crossing each of these points of contact, some of the vibrational energy is dissipated through friction which means that said vibration is then damped quickly and effectively in said structure 1, as is depicted in figure 1, with dampings e1 and e2 for the longitudinal vibrations E and dampings f1 and f2 for the lateral vibrations F.

Of course, the structure 1 may have various shapes, and vary in its degree of massiveness. As a preference, however, it has an elongate shape, in the form bar for example, and said internal cavity 2 is formed longitudinally to said structure 1 inside a tubular chamber 6, as depicted in figure 1.

In the context of the present invention, said solid bodies 9, which are made, for example of synthetic material, preferably beads, may be:

- either compact, their entire mass then being occupied by material;
- or hollow, which makes it possible to reduce the weight of said solid bodies 9 and therefore also the weight of the structure 1.

In addition, according to the invention, said solid bodies 9:

- may be made of different materials (polymer, metallic ceramic, elastomer, etc.), as depicted in figure 2 which shows solid bodies 9A and 9B of identical shape and size, but made of different materials; and/or
- may have different shapes and/or sizes (diameters), as depicted in figure 3, particularly in respect of

the bodies 9C, 9D, 9E and 9F.

It will be noted that:

- a difference in inertia of the solid bodies 9A to 9F, due in particular to different sizes or different densities; and/or
- a difference in stiffness of the solid bodies 9A and 9B, due especially to different materials (for example a material which is not very rigid and intrinsically has very good damping properties or a material which is more rigid and intrinsically has less good damping properties),

give rise to a different movement under the effect of a vibrational excitation and therefore also to a different degree of damping. In consequence, through a suitable selection of these characteristics, it is possible to adjust and to optimize the damping performed by the structure 1.

In addition to said solid bodies 9, which may be compact or hollow, the aggregate 8 may also contain a viscous liquid, for example oil, filling the empty spaces in the chamber 6 between said solid bodies 9. These are then immersed in a lubricating medium, which makes it possible to delay any heating that might occur.

Moreover, in a preferred form of embodiment depicted in figure 2, the means 10 comprise:

- a rigid plate 11, for example a metal plate, which is tailored to the opening 7 so as to be able, preferably with sealing, to close off the chamber 6; and
- an elastic means 12, preferably a spring, which exerts elastic pressure on said rigid plate 11 so as to constrain the aggregate 8, that is to say to press it into the chamber 6, and even possibly to compress it if it comprises a small amount of liquid or if the solid bodies 9 are not very rigid.

Furthermore, the damping structure 1 according to the invention additionally comprises at least one internal partition 13, which is secured to a wall 3 or

4 of the chamber 6 of the structure 1 and which is arranged inside the cavity 2.

By way of illustration, various examples of partition 13 are depicted:

- in a schematic view, in longitudinal section, in figures 4 to 7; and
- in a plan view, in figures 8 to 11.

As can be seen from these figures 4 to 11, the partitions 13:

- may be solid (figures 4, 5, 6, 7, 8, 10 and 11) or pierced (figures 5, 9 and 11); and
- may have any shapes, for example flat (figures 4 to 9) or tubular (figures 10 and 11). In the latter case, the partitions 13 may have any type of cross section; circular, elliptical, or simply any shape.

These internal partitions 13 make it possible to increase the exchange area and therefore the area for friction between, on the one hand, the interior faces of the walls 3, 4 of the chamber 6 and, on the other hand, the aggregate 8, something which makes it possible to increase the damping of the vibration of the structure 1.

In addition to the aforementioned advantages, the structure 1 according to the invention also has the following advantages:

- it can easily be produced and has a low manufacturing cost, particularly when the internal cavity 2 already exists in the structure 1;
- it is of low mass (particularly when use is made of hollow solid bodies 9) by comparison with certain known damping means, such as damping materials bonded directly onto the surface of the structure that is to be damped;
- the aggregate it contains is protected against external attack (fire, moisture, corrosive agents, etc.) by the chamber 6;
- it is effective over a broad band of frequencies and this is true for various types of deformation (bending, tension-compression, torsion, etc.) of the

structure 1;

- it is not subjected to phenomena of abrasion, corrosion or erosion, if an appropriate pair of materials is chosen for the wall 3, 4 of the structure 1 and for the aggregate 8, respectively; and
- it leads to no change in the life of the components with which it is associated.

The physical effect of the filling (of the cavity 2 by the aggregate 8) on the vibrational behavior of an initially hollow structure 1 (the cavity 2 existing, but being empty) will be specified hereinbelow with reference to figure 12.

Three different types of loading of the hollow structures 1 may be dealt with by filling with an aggregate 8, namely:

- bending;
- tension-compression; and
- torsion.

The vibrational response of a hollow structure 1 is considered as the linear superposition of second-order system responses, each characterized by a natural frequency, modal damping, a modal mass and a modal stiffness.

At a given frequency, the structure 1 and the collection of aggregate 8 may be replaced by two coupled systems depicted in figure 12, in which:

- MA and KA represent, respectively, the true modal mass and the true modal stiffness of the untreated structure 1 loaded in bending, longitudinal or torsion;
- MB represents the equivalent mass of the aggregate 8, set in motion by the coupling with the hollow structure 1 loaded in bending, longitudinal or torsion; and
- CB indicates the internal friction provided by the aggregate 8.

The filling of the cavity 2 modifies the vibrational response of the structure 1 but does not

modify the excitation force F0 originating from the excitation upstream (the housing, for example, in the case of a helicopter main gearbox).

Under harmonic conditions, the respective displacements during the time $x1(t)$ and $x2(t)$, the respective velocities $v1(t)$ and $v2(t)$ and the respective accelerations $\gamma 1(t)$ and $\gamma 2(t)$ satisfy, for any angular frequency ω of the excitation force of amplitude F, where $F0(t) = F(\omega).\sin(\omega t)$:

$v1(t) = j\omega x1(t)$ and $v(2)t = j\omega x2(t)$

$\gamma 1(t) = -\omega^2 x1(t)$ and $\gamma 2 = -\omega^2 x2(t)$.

The sum of the applied forces (return forces, friction force due to the coupling with the other mass, and possibly external force F0) for each mass being equal to its inertial force, it is therefore possible to write, for each mass:

- as a function of time <u>t</u>:
  - in the case of the mass MA: $F0(t) - KAx1(t) - CB(v1(t)-v2(t)) = MA\gamma 1(t)$
  - in the case of the mass MB: $0 - CB(v2(t)-v1(t)) = MB\gamma 2(t)$
- as a function of angular frequency ω:
  - in the case of the mass MA: $F(\omega) - KAX1(\omega) - CBj\omega(X1(\omega)-X2(\omega)) = -MA\omega^2 X1(\omega)$
  - in the case of the mass MB: $0 - CBj\omega(X2(\omega)-X1(\omega)) = -MB\omega^2 X2(\omega)$

with $j^2 = -1$ and $X1(\omega)$ and $X2(\omega)$ being complex quantities.

From there, it is easy to determine (by considering the frequency f in Hz) the force/acceleration amplitude spectrum and the spectrum of the phase shift of the acceleration with respect to force, these being accessible through measurement (with $f = \omega/2\pi$ and $fA = \omega A/2\pi$, fA and ωA being, respectively, the natural frequency and the natural angular frequency of the structure A (unfilled structure 1)).

From that, it can be deduced that the effect of the filling (aggregate 8) of the chamber 6 on the vibrational behavior of the structure 1 results in:

- a strong decrease in the maximum amplitude response (defining the resonant frequency of the damped system);
- a relatively significant slide in the maximum of the

amplitude response toward the low frequencies;

- a significant widening of the amplitude response spectrum; and
- a significant flattening of the phase response curve.

Furthermore, the coefficient CB may be expressed theoretically near the natural mode ωA by:

$CB = \alpha B\ 2\pi fA\ mB\ \tan(\delta B)$

with:

- αB: a dimensionless coefficient which indicates the actual effectiveness of the filling;
- δB: an intrinsic loss angle for the filling material, known beforehand; and
- mB: the physical mass added by the filling (aggregate 8).

It is therefore found to be true that:

- the greater the material loss angle, the greater the equivalent damping;
- the equivalent damping is proportional to mB; and
- the better the quality of contact, the better the damping efficiency of the filling (aggregate 8).

Optimizing the damping consists in increasing CB, that is to say the damping mass of the aggregate 8, which is defined by αB mB tan(δB).

The technological parameters that allow this damping mass to be increased are:

- in the case of the loss angle δB:
  - the number of types of solid bodies 9 used (one single type or a mixture of several types);
  - the nature of the constituents: polymer, metallic ceramic or elastomer;
  - the viscosity of any filling liquid used,
- in the case of the effectiveness coefficient αB:
  - the surface finish of the solid bodies 9 that make up the aggregate 8;
  - the static compacting pressure generated by the means 10,
- in the case of the mass of filling mB:
  - the mean volumetric mass of the solid bodies 9 of

which the aggregate 8 is made;

- the mean diameter of the solid bodies 9 of which the aggregate 8 is made;
- the wall thickness of the solid bodies 9 of which the aggregate 8 is made, if the bodies are hollow.

Numerous applications are of course possible for the damping structure 1 according to the invention.

In particular, said structure 1 can be used to damp vibration of various types of vibrating component. It may thus, in particular, be employed as part:

- of a connecting bar connecting between a support that is to be isolated with respect to vibration and a casing surrounding rotating parts generating this vibration, as will be seen in greater detail hereinafter with reference to figure 13; or
- of the suspension for an engine, a gearbox or a rotating part, such as a compressor or a fan, for example.

According to the invention, in order to suspend a vibrating component with respect to a support in such a way as to isolate the latter from the vibration of said vibrating component, one or more structures 1, particularly in the form of bars, may:

- be attached and arranged at free points between the vibrating component and the support; or
- replace elements, for example connecting rods, which already exist on the component or the support; or
- be formed in elements (hollow or otherwise) which already exist.

The last two solutions additionally have the advantage of not adding to the bulk.

Preferred applications of the damping structure 1 relate to the reduction of noise-generating vibration in a rotary-wing aircraft, particularly a helicopter, and in particular to reducing:

- the running or meshing noise from gearboxes; and/or
- the running or meshing noise of auxiliary units (lubricating pumps, drive of ventilating and air-

conditioning units, etc.),

which noise is very annoying in the cabin, both for the pilots and the passengers.

The particular application of the invention depicted in figure 13 is intended to increase the damping of suspension bars 15 of a system used for the suspension of the main gearbox BTP (connected to the mast 16 of the lift and forward travel rotor) of a helicopter He, which suspension bars 15 are arranged on the fuselage 17 of the helicopter He.

To do this, the suspension bars 15 each comprise a damping structure 1 according to the invention, as can be seen in the case of one of these bars 15 which is partially cut away in figure 13.

This is done in order to reduce, in the cabin, the noise propagated through solid transmitted by the bars 15, that is to say the vibrational energy transmitted by said bars 15, indicated by an expression $|H(f)||\gamma bar|^2(f)$ specified hereinbelow.

In general, it may be considered that the acoustic pressure spectrum in the cabin of the helicopter He, denoted Pcab(f), satisfies the following quadratic relationship:

$$Pcab^2(f)=|T(f)||Pdirect|^2(f)+|H(f)||\gamma bar|^2(f)+|Q(f)||\gamma structure|^2(f)$$

Specifically, this summing of squared amplitudes indicates the balance of energy transfers for the meshing noise at frequencies higher than 500 Hz. There is no need to take account of the phase relationships between the pressure in the cabin and the direct pressure or accelerations of the structure (fuselage) of the helicopter He, given the great many acoustic modes present in the cabin at these frequencies.

It will be noted that:

- the term $|T(f)||Pdirect|^2(f)$ represents the quadratic acoustic pressure in the cabin, due solely to the noise radiated directly by the main gearbox BTP of the helicopter He. $|T(f)|$ represents the modulus of the coefficient of acoustic transmission

(dimensionless) of the noise radiated through the air of amplitude Pdirect as far as the cabin;

- the term $|H(f)||\gamma bar|^2(f)$ represents the acoustic pressure in the cabin, due solely to the noise radiated in the cabin by the structure (part of the fuselage 17) excited by the vibration of the attachments of the bars 15. $|H(f)|$ represents the effectiveness of acoustic radiation in the cabin of the vibration of this part of the fuselage;
- the term $|Q(f)||\gamma structure|^2(f)$ represents the quadratic acoustic pressure in the cabin due solely to the noise radiated in the cabin by the remainder of the fuselage which is not excited by the vibration of the attachments of the bars 15, but by the bottom of the gearbox BTP for example. $|Q(f)|$ represents the modulus of the coefficient of acoustic radiation in the cabin of this last part of the fuselage.

From the foregoing, it would seem that the reduction of the noise in the cabin will be significant at the meshing frequencies for which the following relationship is satisfied in the absence of treatment:

$$|H(f)||\gamma bar|^2(f) >> |T(f)||Pdirect|^2(f) + |Q(f)||\gamma structure|^2(f).$$